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# AERODYNAMICS OF AIR CUSHION VEHICLES

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## CUSHIONS, FAN AND DUCTING SYSTEMS

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## 1. Introductory Remarks

In a comparatively short period of time it is clearly impossible to cover all the aspects of the A.C.V. cushion and its associated systems and hence in this lecture we can do little more than assess the overall picture. Two main references which have been used in the preparation of the lecture may prove useful to those intending to study A.C.V.'s in more detail are

a) Hovercraft Design and Construction
by G. H. Elsley and A. J. Devereux,
published by David and Charles
Newton Abbot 1968

and b) A Literature Survey on the Aerodynamics of Air Cushion Vehicles
by A. Harting
Agard Report 565, 1969.

## 2. The Unskirted Cushion

### i) Plenum Type

The plenum type is perhaps the simplest of the many types of A.C.V. that have been built and at this point in time most operational craft utilise this principle which has largely superseeded the peripheral jet for reasons which will be discussed later. Figure 1 shows a section through an unskirted plenum craft. At lift off air is pumped into the cavity by a suitable compressor and diffuses to form the cushion. The craft rises until a steady state condition is achieved in which the incoming air matches exactly the air being lost to atmosphere at the periphery.

Assuming that the air within the cushion is at rest we may write using Bernoulli's theorem

$$V_c = \left(\frac{2}{\rho} p_c\right)^{\frac{1}{2}} \qquad \ldots\ldots\ldots(1)$$

where $V_c$ = cushion escape velocity

$\rho$ = density of the air

$p_c$ = cushion pressure (relative to atmosphere)

The total volume flow of air is then given by

$$\mathcal{V} = V_c hCD_c = \left(\frac{2}{\rho} p_c\right)^{\frac{1}{2}} hCD_c \qquad \text{........(2)}$$

where h = air gap or clearance height

$D_c$ = discharge coefficient

C = cushion perimeter

$\mathcal{V}$ = volume flow

The discharge coefficient, which may be defined as the ratio of the actual volume flow to that calculated assuming the gap ran full at the cushion escape velocity i.e.

$$D_c = \frac{\mathcal{V}}{hCV_c} \qquad \text{........(3)}$$

is a function of the angle $\theta$ and the length of the wall. For a long wall theoretical non viscous values of $D_c$ are:-

| | | | | | |
|---|---|---|---|---|---|
| $\theta$ | 0 | 45 | 90 | 135 | 180 |
| $D_c$ | 0.500 | 0.537 | 0.611 | 0.746 | 1.000 |

The power required at the peripheral gap, i.e. neglecting all losses, is given by

$$P = p_c \mathcal{V} \qquad \text{........(4)}$$

or

$$P = p_c^{3/2}\left(\frac{2}{\rho}\right)^{\frac{1}{2}} hCD_c \qquad \text{........(5)}$$

writing $p_c = \frac{W}{S}$ ........(6)

where W = craft weight

S = cushion area

equation 6 may be rewritten as

$$P = \left(\frac{W}{S}\right)^{3/2}\left(\frac{2}{\rho}\right)^{\frac{1}{2}} hCD_c \qquad \text{........(7)}$$

Useful conclusions as to the variation of air gap power or what is sometimes called 'ideal lift power' with weight, clearance, etc. can be drawn from this equation.

## ii) Peripheral Jet Type

There have been almost as many peripheral jet theories as there have been engineers and applied mathematicians working in the A.C.V. field, as a study of the references will show, ranging from simple momentum through complex transformation to elaborate mixing theories. A limited selection are discussed below.

### a) Simple Momentum Theory

Figure 2 represents a section through an unskirted peripheral jet craft. Air is pumped through the nozzle and generates a pressure under the craft which, as with a plenum chamber system lifts it to some nominal clearance height h. At this height we assume that there is no mixing or entrainment and hence that no air enters or leaves the cushion. This implies that the jet is bent round and becomes tangential to the ground.

We assume a thin jet and conservation of momentum along the jet and hence can write

$$p_c h = J(1 + \cos \theta) \qquad \text{........(8)}$$

where J = nozzle momentum per unit periphery.

We further assume that the jet total pressure $H_J$, and the nozzle static pressure p are constant across the jet and that $p = kp_c$ where k is some constant.

The jet velocity $$V_J = \left(\frac{2}{\rho}\left\{H_J - kp_c\right\}\right)^{\frac{1}{2}} \qquad \text{........(9)}$$

and since $$J = \rho V_J^2 t \qquad \text{........(10)}$$

$$p_c h = 2\left\{H_J - kp_c\right\} t\,(1 + \cos \theta) \qquad \text{.......(11)}$$

rearranging and writing $x = \frac{t}{h}(1 + \cos \theta)$ ........(12)

leads to $$\frac{p_c}{H_J} = \frac{2x}{1 + 2kx} \qquad \text{........(13)}$$

Noting that $\nu = V_J Ct$ ........(14)

leads to

$$\nu = \left[\frac{2}{\rho} H_J \left\{\frac{1}{1 + 2kx}\right\}\right]^{\frac{1}{2}} Ct \qquad ........(15)$$

An additional lift term derives from the jet reaction $R_J$

$$R_J = (kp_c + \rho V_J^2)t \text{ per unit length} \quad ...(16)$$

or

$$R_J = 2H_J \left\{\frac{1 + kx}{1 + 2kx}\right\} t \qquad ........(17)$$

The ideal total or nozzle power is given by

$$P = H_J \nu = H_J^{3/2} \left[\frac{2}{\rho} \left\{\frac{1}{1 + 2kx}\right\}\right]^{\frac{1}{2}} Ct \qquad ........(18)$$

and defining the cushion area to the nozzle outer wall the total lift is

$$L = p_c S + R_J C \sin \theta - p_c Ct \sin \theta \qquad ........(19)$$

To use these equations it remains to determine k the ratio of jet static pressure to cushion pressure. Reference to equation (13) shows that as $x \to \infty$ $\frac{p_c}{H_J}$ should $\to 1$ this is given by $k = 1$ however this leads to low cushion pressures at low values of x. At Cowes we used $k = 0.5$ i.e. the jet static pressure a mean of cushion and atmosphere. This gave incorrect results at high x but reasonably close agreement throughout the rest of the range.

b) Other Jet Theories

Table I shows the results of some other relatively simple non mixing theories. Of these the most accurate is that due independently to Wald, Thunholm and Hughes. At Cowes we tended to use the 'exponential' theory. While not particularly rigorous mathematically it was easy to use and gave results, for jet widths above about 1 inch, in reasonable agreement with practice.

Using any of these theories it is possible to manipulate the equations and derive for example the optimum value of x or jet width. Including the power required to overcome momentum drag, which tends to reduce jet width as craft speed increases gives additional scope for manipulation as did the proposed use of angled vanes in the jet to provide propulsion. Most of this work is now only of historical interest but the one time emphasis on the peripheral jet can be partly understood by utilizing the preceding equations to form the ratio

$$\frac{\text{Ideal Jet Power}}{\text{Ideal Plenum Power}}$$

for a given cushion area, weight and clearance height. Typically the peripheral jet requires only 70% of the plenum types power.

## 3. The Skirt Cushion and its relation to the Skirt System

In the previous section it was seen that in the ideal overland case the peripheral jet craft required less power than the plenum craft. Following this argument early non skirted craft, typically those built by B.H.C. and its predecessors, were often of the peripheral jet type. Because, as is usually the case, practice never reached the theoretical best clearances were lower than expected and ways were sought of a) improving the effective clearance and b) reducing the lift power to do this. On both plenum craft (Curtiss Wright Aerocar) and peripheral jet craft (S.R.N1) this lead directly to some form of skirt. With plenum craft this led initially to a single flexible extension to the cavity wall restrained by ties or occasionally some brush like device while on peripheral jetted craft a double walled extension of the nozzle appeared.

Clearly it was possible to calculate the plenum type performance much as discussed previously but difficulties arose in the peripheral jet case associated with maintaining the jet width and angle. This was further complicated by the difficulty of designing an effective non scooping jet extension across the rear of the craft. This initially led to the fitting of sealed bags at the rear of the craft which tended to negate the peripheral jet principle, since they had to operate in the inefficient split condition to feed the rear gap. Again the split condition occurred with operation over waves and undulating surfaces. Thus although in theory the peripheral jet was still superior to the plenum in practice this was probably not so.

With the relatively 'hard' nozzle of the flexible extensions, e.g. that originally fitted to SR.N5's and SR.N6's, considerable wear took place and it was this wear problem that led to the separate finger solution and the essentially plenum chamber configuration of present SR.N-B.H.C. series craft. On these craft the air supply to the cushion is through two sets of discrete holes, from bag to finger and from bag to cushion.

From the air gap power point of view there is no difference between this type of craft and the basic plenum craft discussed in section 2 except in that the air gap tends to be very much smaller with the skirt. There are however some additional losses associated with the requirement to maintain a significant bag pressure/cushion pressure ratio and an effect on craft response associated with the much greater depth of cushion.

## 4. Internal Aerodynamics

### Possible choices of fan installation

It is assumed that it is required to feed a peripheral bag system such as used in BHC craft, for reasons discussed in the lecture on skirt design.

Transmission simplicity requires a minimum number of fans and each must therefore supply a fairly long portion of the periphery. This generally results in a high rate diffusion.

Symmetry of installation is usually desirable from point of view of standardisation of structural components - bulkheads, frames, etc., so that single fans are usually placed on the longitudinal centre line and multiple fans symmetrically disposed relative to the longitudinal centre-line. All-round visibility, low profile, clear payload space usually mean fan discs horizontal, and no large ducts above floor.

One is then faced with a comparative installation as shown in fig. 3 and it is generally apparent that the axial fan will still result in a higher profile and C.G. Axial fans are generally more sensitive to poor intake distribution due to intake obstructions.

Although with an axial fan the total efficiency for given flow and pressure at section B B could be marginally higher than generally achievable with a centrifugal fan, the latter has the advantage of delivery at section A A and with careful design, a net overall gain is likely to result.

A further argument may be levelled at the axial fan if an integrated lift/propulsion system is not used. This is that single stage axial fans may have a power characteristic which rises as the flow falls below the working point. This means that the power requirement is a maximum at zero flow. Therefore, unless more power is available to the fan than the normal power, as would be the case with an integrated system, some kind of blow-off device would be necessary to achieve lift off. With the backward curved blade centrifugal fan, the power requirement is low at shut-off and rises through the working point.

Another advantage of the centrifugal fan in A.C.V. applications is the general ruggedness and simplicity of design, which make the fan more tolerant of the environment.

(a) Fan Geometry and Performance

These aspects are intimately linked and will be considered in the light of BHC experience with centrifugal fans.

The fans used in BHC craft were originally chosen on a basis of high total efficiency and the fact that they were generally geometrically compatible with craft layout under the required working conditions.

One of the factors generally taken into account when selecting fan type, is the specific speed. This is defined as

$$N_S = N \frac{v^{1/2}}{H^{3/4}} \tag{20}$$

where, for compressors N is generally in r.p.m., v in cusec and H in ft.

Replacing N by tip speed and fan diameter gives

$$N_S = \frac{60}{\pi} \frac{V}{D} \frac{v^{1/2}}{H^{3/4}} \tag{21}$$

where V is in ft/sec and D in ft.

Considerations of noise emission suggest that a tip speed of 300 ft/sec is a reasonable upper limit, and accepting for the moment that free diffusion is a relatively inefficient process, it is desirable to choose D as large as possible. When pressure/flow requirements and the diameter limitation for typical craft (N4, N6, BH.7) are inserted in the equation, the value of specific speed is found to be of order 200. Various workers have produced envelope curves of efficiency for the different types of impeller as functions of specific speed and these suggest that a mixed flow impeller would normally prove to be most efficient for this particular specific speed. Such an impeller was in fact designed by BHC/NEL, ref. 1, and model tests were very encouraging. However, manufacture of this type of impeller is quite difficult since the blade form is generally complex. For ease of construction therefore the standard single curvature, backward curved blade design, has been used by BHC. Typical performance curves, for the SR85 fan, are shown in fig. 4. The head and flow coefficients used are not non-dimensional, but are analogous to the standard coefficients $\psi$ and $\phi$ respectively. Generally convenient units were used, i.e. fan total pressure rise H in lb/ft$^2$, volume flow v in cusec, and rotational speed N in r.p.m. The total efficiency shown is rather low, but this was expected since the tests were performed on a small (1 ft dia.) impeller. At larger sizes a geometrically similar fan should achieve 85%.

Also shown in the slide are typical curves obtained for impellers of rather larger exit depth/diameter ratio (0.3 approx.), type G1, described in ref. 2. The impeller diameter in this case was 28 ins. and the total efficiency is improved partly because of this scale effect.

The main virtue of this deeper fan lies in the fact that the static efficiency is higher, i.e. more of the energy is produced as static pressure rise. If the extra depth can be accommodated structurally, the overall system efficiency is then significantly improved.

In order to maintain satisfactory flow conditions at the impeller top disc, the blades in this design are twisted. Again, this results in greater manufacturing difficulty although the blades can be designed to have only single curvature.

The major quasi non-dimensional parameters for fan performance, obtained by using the techniques of dimensional analysis are $\frac{H}{N^2D^2}$ and $\frac{V}{ND^3}$, but a further parameter, $\frac{HD^4}{V^2}$ may be derived. This parameter is particularly useful when plotted against $\frac{V}{ND^3}$. At an early stage in the craft design the cushion pressure and flow requirement will have been specified. This allows some estimation of the pressure loss and hence determination of H. $\frac{HD^4}{V^2}$ may then be derived for the available D or range of D and use of the curve gives the required fan speed. If a series of curves is drawn for different fan designs, it is usually possible to ensure that the working point coincides with the point of maximum impeller efficiency.

Historically, the BHC fans were used first in the twin-fan craft SR.N2 and SR.N3, with a shallow exit depth. In the next craft, SR.N5, a relatively greater flow was required for the given diameter and the fan depth diameter was therefore increased to 0.233. This modification was expected to increase static efficiency and reduce losses downstream of the fan. In fact, model tests suggested that there was some flow separation at the top disc and total efficiency was slightly reduced. Subsequent fan depth/diameter ratios have therefore been kept below a value of 0.2, which has proved generally satisfactory. The expansion ratio from eye to fan periphery with this depth/diameter ratio is 2.0, a value generally considered to be a maximum for this type of fan. Whether or not the system working point is chosen to match the fan maximum total or static efficiency point really depends upon the diffusion efficiency, but generally the two points are sufficiently close for the matching not to be critical.

## Remarks on Dynamic Aspects

### Characteristic Slope

Although fan type and performance characteristic generally have been selected on a basis of static performance, current investigations into craft motion suggest that more emphasis may have to be placed on dynamic aspects. This is diagrammatically illustrated in fig. 5. Here, two fans with different characteristic slopes are shown, having the same static working point, A.

Due to a flow reduction $\Delta v$ (due to change of craft position relative to the surface) the working point will move to B and C. If the pressure loss/flow relationship from fan to cushion is fixed, the cushion pressure will rise from the initial value D to a new value at E or F depending on the fan characteristic. Thus in effect the cushion spring stiffness may be controlled by choice of fan characteristic slope. It is interesting to note that a positive stiffness is obtained even when the fan characteristic slope is zero. This illustration assumes that fan speed remains constant during the working point excursions. Experience with BHC craft substantiates this assumption.

Parallel Operation

A type of flow instability may result due to dynamic effects. This would apply in cases where several fans are mounted in parallel and the characteristic has a region of positive slope. If the working point is close to the peak in the characteristic then it is possible for some fans to work below the peak and others above, for the same pressure requirement. Changes in flow requirement may then drive some fans towards shut off and the remainder to higher flow. This condition can be guarded against by using fans with completely negative slope. Experience with SR.N4 which has four fans in parallel, each with a positive slope in the characteristic, suggests however that this is not a serious problem.

(b) Ducting

As mentioned earlier, the general requirement to make maximum use of the payload volume suggests that most of the discharge ducting should be below cabin floor level and that intake length should be as short as possible.

Intakes

In the ACV application, a primary requirement is that performance should not be greatly affected when operating at large aerodynamic yaw angles. In addition, when using air propulsion and an integrated lift/propulsion system, it is generally desirable to mount the propellers close to the fans in order to keep shaft lengths short and the number of gearboxes to a minimum. For mechanical reasons then, we have found it most convenient to mount propellers on pylons directly above the fan intakes, with provision to rotate the pylons for thrust vectoring. The above considerations make it extremely difficult and probably undesirable to design the intake to recover the dynamic pressure due to relative air-speed. Model tests have shown however that providing there is a reasonably unobstructed length of roof for some six fan diameters ahead of the intake, recoveries of up to 0.7 times dynamic pressure may be possible, but the situation may be complicated by having intakes and propellers in line so that the rear intake swallows some of the forward propeller slipstream.

Static measurements show that the major source of loss is the beam on which the pylon structure is mounted. This leads to uneven velocity distributions at the eye of the fan, which have not been improved significantly by fairing the beam in various ways.

<u>Diffusion</u>

Although the diffusion process in the plenum chamber is often termed 'free' diffusion, the actual situation is often far from ideal. In a practical design, the diffuser exit is usually straight sided so that the diffusion path length is not constant. Many obstructions (roof supports, cabin walls, etc.) are found in the flow path. Our own model tests have shown however that the usual support tubes cause little trouble unless mounted very close to the impeller exit. Work at N.E.L., ref. 3, has shown that if a vertical, flat plate is mounted in this type of diffuser, further from the impeller centre-line than 80% of the plenum exit diameter, a marginal improvement in performance may be obtained.

The pressure loss between the fan exit and the bag can be found as a function of the whirl and radial velocity at the fan exit i.e.

$$\Delta p = \frac{1}{2} \rho C (V_r^2 + V_w^2) \qquad (22)$$

where $\rho$ is atmospheric density

$C$ is an empirical constant

$V_r$ is the fan radial velocity

$V_w$ is the air whirl velocity.

However, if the fan parameter $v/ND^3$ reduces below the optimum value, the whirl velocity component increases even though the radial velocity component decreases. Thus the pressure loss calculated from this expression tends to be artificially high at low volume flows.

A correction for this can be made by allowing for the expansion of the air as it moves away from the fan. This allowance is made by including the direction of the air relative to the fan periphery such that

$$\Delta p = \frac{1}{2} \rho C (V_r^2 + V_w^2) \tan^{-1}\left(\frac{V_r}{V_w}\right) \qquad (23)$$

The pressure loss will now be found to reduce to zero as the volume flow reduces to zero and this agrees more closely with experimental observation.

The air velocities at entry to the peripheral bags are usually quite low (of order 50 ft/sec) so that further bend and expansion losses are also low and do not warrant the fitting of fairings or turning vanes.

The final pressure loss in the system occurs across the feedholes between the bag and cushion and is given by the standard orifice equation

$$v = D_c A \sqrt{\frac{2}{\rho} \Delta p} \qquad (24)$$

where $v$ is the volume flow

$A$ the feedhole area

$D_c$ the discharge coefficient

$\Delta p$ the pressure differential across the holes

$\rho$ the air density.

Normally, with a sharp-edged orifice which is plane and perpendicular to the airflow, we have $D_c$ of order 0.61. There is however a possibility that when the orifice is formed of flexible material, the edge may be deformed in relation to the pressure differential. This would result in the relationship between $v$ and $\Delta p$ being more nearly linear.

To summarise, a typical pressure drop distribution for BHC craft of the bag/fingered skirt type would be :

| | |
|---|---|
| cushion | 40% |
| bag/cushion | 20% |
| plenum | 20% |
| intake | 20% |

Thus, if lift efficiency is defined as

$$\frac{\text{cushion pressure}}{\text{fan total pressure}} \times \text{fan total efficiency}$$

one rarely achieves values better than 35%.

## Flow losses

In addition to the pressure losses just discussed, there may be flow losses. These can take place through the hinges connecting the flexible skirt to the hard structure, or they may be 'useful' leaks such as the rudder control ducts on SR.N5 and SR.N6. Control ports are another source of intermittent flow loss which can be a large fraction of the total flow. A contingency allowance must therefore be added to flow calculations in the initial design stages.

One further source of flow loss is the engine air flow. Engine specifications often quote salt concentrations as low as 0.01 parts per million. This requires very high filtration efficiency indeed and is very difficult to achieve in the region of a spray generator such as an A.C.V. One partial solution developed at B.H.C. has been to draw the engine air from the plenum chamber, via various water strippers and filter panels of the 'Knitmesh' variety. Such arrangements have proved moderately successful but do require further plenum obstructions and for example in SR.N4 some 10% of the cushion flow. The main point of taking the air from the plenum chamber is that the fan tends to coalesce the spray and by suitable choice of extraction point, usually the plenum chamber roof, the air is relatively dry to start with.

## 5. Performance

### i) Lift Power Requirements

As discussed in section 3 the ideal lift power requirement of the plenum type craft is given by

$$P = \upsilon p_c \quad \text{in the static case}$$

including the loss terms treated previously and any ram recovery due to fuel speed

$$P = \upsilon\Big(p_c + \text{intake loss}\left\{f_1(\upsilon^2)\right\} + \text{diff. loss}\left\{f_2(\upsilon^2, \text{rpm})\right\}$$
$$+ \text{bag cushion loss}\left\{f_3(\upsilon^2)\right\} - k\tfrac{1}{2}\rho V_o^{\,2}\Big)\eta_F \quad \text{........(25)}$$

where $\upsilon$ is taken to be the total volume flow including any leaks propulsion or control bleeds and $\eta_F$ is the fan efficiency.

In general the cushion pressure is known from the craft weight and cushion area and a preliminary assessment of the required lift volume flow can be made from clearance considerations. Typically skirted craft operate at clearance/length ratios ($^h/_\ell$) in the range 0.003 $\rightarrow$ 0.005. An alternative presentation of this data is as $C_Q$ where

$$C_Q = \frac{Q}{S(\frac{2}{\rho}p_c)^{\frac{1}{2}}} \qquad \text{and } Q \equiv \upsilon \quad \text{........(26)}$$

this is equivalent to $C_Q \propto \frac{h}{\ell}$

| | | | |
|---|---|---|---|
| typical $C_Q$ values are | 0.015 | 0.030 | amphibious craft |
| | 0.005 | 0.010 | sidewall craft. |

The addition of leakage and bleed volume flow enables an initial calculation of the system losses and hence fan operating point to be made. Because the diffusion losses are in general a function of fan rpm as well as of volume flow the calculation is usually iterative. Typical values of lift horse power per ton range from about 10 up to about 40 depending on craft size, type and application.

ii) Resistance of the Air Cushion System

Apart from the resistance terms associated with the craft skirts, contact and profile terms, three major resistance terms derive directly from the presence of the air cushion system.

a) Momentum Drag

This arises from the fact that the air supply must be accelerated from rest up to the craft's relative air speed. This may be written $D_m = \rho v V_o$

The net effect of this will be modified if the air escape from the cushion, leaks etc. is not uniform around the craft and some interference terms may arise if intakes and propulsive devices are in close proximity.

b) Trim Drag

This arises from an inclination of the lift vector if the skirt hemline, or cushion base with an unskirted craft, is not horizontal. Clearly this can be either a thrust or a drag term depending on which way the craft is trimmed.

c) Wave Making Drag

At zero forward speed over water the cushion pressure creates a depression of the surface beneath the craft of depth

$$d = \frac{p_c}{\rho_w g} \qquad \text{........(27)}$$

At forward speed the cushion generates waves the energy loss of which can be considered as corresponding to the wave making drag $D_w$.

Fig. 6 , based on a figure from Lamb's Hydrodynamics, shows the wave profile for a cushion length

$$\ell = \frac{V_w^{\,2}}{g} \quad \text{(Froude No. = 1)}$$

From Lamb's results Crewe and Eggington derived the wave making drag as

$$\frac{D_w}{L} = \frac{h}{\ell} = \frac{2p_c}{\ell \rho_w g}\left(1 - \cos\frac{g\ell}{V_w^{\,2}}\right)$$

which has been plotted in fig. 7 as

$$\frac{D_w}{p_c^{\,2} S} = \frac{2}{\rho_w g}\left(1 - \cos\frac{g}{V_w^{\,2}}\right) \qquad \ldots\ldots\ldots(28)$$

Since this relatively simple solution was prepared other workers notably Newman and Pook, and Barrat have produced solutions applicable to a whole range of craft planforms and water depths.

iii) Use of Cushion for Control

Among ways of using the cushion for control purposes may be mentioned a) Physically moving the cushion and b) Modifying the pressure distribution.

a) Moving the Cushion

This method was tried on a variant of SR.N1 in which the peripheral jet was switched between two positions. More recently the lower edge of the skirt has been moved in a horizontal direction by means of jacks and cables on both H.D.2. and V.T.1. Both these systems moved the cushion, and hence its centre of pressure, relative to the craft centre of gravity thus giving rise to rolling or pitching moments.

b) Modifying the Pressure Distribution

Unlike the previous method this relies on some degree of cushion compartmentation to sustain a pressure differential. Essentially it is achieved by varying the supply of air to or escape of air from one or more of the cushion compartments. Early examples of this were the roll control values on SR.N1 and the skirt life used on SR.N5 (and more recently on BH.7) where the cushion supply was locally stopped or the cushion allowed to escape respectively. Both of these methods modified the pressure distribution and hence gave rise to a moment about the craft c.g.

Clearly with a compartmental cushion and separate fans a similar result, albeit more of a trim than a control device, can be obtained by differential fan rpm. Again valving the cushion to atmosphere will give a similar result and the discharge can be vectored.

One possible advantage or perhaps disadvantage of operating on the cushion is that it can lead to excessive skirt immersion and hence generate a substantial yawing moment.

## 6. Cushion Response

In any determination of A.C.V. response, there are two major problem areas

(a) the complexity of the forcing

(b) the non-linearity of the system.

With regard to (a), it may be noted that sea waves, when regular, are well described by the trochoidal form and have length to height ratios from 10 upwards.

The parametric equations of the trochoid are

$$x = \frac{\lambda'}{2\pi} \cdot \theta + r \sin \theta \tag{29}$$

$$y = \frac{\lambda'}{2\pi} + r \cos \theta \tag{30}$$

where x and y are displacements measured horizontally and downwards, $\lambda$ and r are wave length and amplitude (half height) and $\theta$ is the angle through which the radius of the generating circle has rotated.

The waves however are not generally regular in coastal waters, but this does not mean that they are random. Generally, one finds at least one pronounced peak when spectral energy is plotted against frequency.

These difficulties make the determination of an analytical solution impossible and various types of modelling are therefore used. These may be digital, requiring the use of a computer, or analogue, using either a physical or mathematical model. All of these have been used at BHC and will be discussed later but we will first consider a fairly simple treatment of the problem.

## Linear Theories

The most general assumptions take the craft motion to be represented adequately by a second order linear differential equation with constant coefficients i.e.

$$I_s\ddot{S} + N_s\dot{S} + K_s S = \exp(iwt) \sum_{n=1}^{\infty} \bar{F}_n \exp(-i\delta_n) \quad (31)$$

where $S$ is a generalised displacement (heave, pitch, etc.)

$w$ is the frequency of wave encounter

$\delta_n$ is the phase shift of the $n^{th}$ force

$I_s$ the inertia

$K_s$ the stiffness

$N_s$ the damping

$\bar{F}_n$ the $n^{th}$ force

The amplitude of the forcing function is written as the sum of n components with n phase shifts so that by imposing conditions on $\bar{F}_n$ $\delta_n$ and n, various types of motion may be considered.

Putting $a_n = \dfrac{\bar{F}_n}{I_s}$ and non-dimensionalising, (32)

the coefficients one obtains:

$$\ddot{S} + 2\sigma w_s\dot{S} + w_s^2 S = \exp(iwt) \sum_{n=1}^{\infty} a_n \exp(-i\delta_n) \quad (33)$$

where $\sigma$ is the ratio of damping coefficient to critical damping coefficient

$$\frac{\nu}{\nu_c} = \frac{1}{2}\sqrt{\frac{N_s^2}{K_s I_s}} \quad (34)$$

and $w_s$ = the undamped natural frequency.

The forced motions in a seaway are obtained as particular solutions to equation (33).

Writing

tuning factor $\Lambda = \frac{w}{w_s}$ (35)

and $S = \bar{S} \exp\left[i(wt - \varepsilon)\right]$ (36)

we have

$$\bar{S} = \left[\sum_{n=1}^{\infty} a_n^2\right]^{1/2} w_s^{-2} \left[(1 - \Lambda^2)^2 + (2\sigma\Lambda)^2\right]^{-1/2} \tag{37}$$

and the phase shift of the motion is given by

$$\varepsilon = \tan^{-1}\left(\frac{2\sigma\Lambda}{1 - \Lambda^2}\right) + \tan^{-1} \frac{\sum_{n=1}^{\infty} a_n \sin \delta_n}{\sum_{n=1}^{\infty} a_n \cos \delta_n} \tag{38}$$

The term $\left[\sum_{n=1}^{\infty} a_n^2\right]^{1/2} w_s^{-2}$ represents the static deflection under load and the term $\left[(1 - \Lambda^2)^2 + (2\sigma\Lambda)^2\right]^{-1/2}$, the magnification factor. One may simplify this general treatment further and arrive at two special cases:

(i) <u>Single exciting force</u>

This requires n = 1 and $\delta_1 = 0$ and hence

$$\bar{S} = a_1 w_s^{-2} \left[(1 - \Lambda^2)^2 + (2\sigma\Lambda)^2\right]^{-1/2} \tag{39}$$

and

$$\varepsilon = \tan^{-1} \frac{2\sigma\Lambda}{1 - \Lambda^2} \tag{40}$$

These equations apply to the single spring/dashpot system, where the dashpot damping is relative to the fixed datum.

The magnification factor

$$\mu_s = \left[(1 - \Lambda^2)^2 + (2\sigma\Lambda)^2\right]^{-1/2} \tag{41}$$

and the form of this function and the phase shift of the motion, ε, are shown in fig. 8.

(ii) Two Force System (Relative Damping)

For this, we put $n = 2$ $\delta_1 = 0$, $\delta_2 = \frac{\pi}{2}$

and $a_2 = -2\sigma\Lambda a_1$ .

There are thus two exciting forces, $90^\circ$ out of phase with each other. The final condition ensures that the damping term relates to the velocity of the craft relative to the forcing surface. Here,

$$\bar{S} = a_1 w_s^{-2} \left[ \frac{1 + 4\sigma^2\Lambda^2}{(1-\Lambda^2)^2 + (2\sigma\Lambda)^2} \right]^{1/2} \tag{42}$$

and hence

$$\mu_s = \left[ \frac{1 + 4\sigma^2\Lambda^2}{(1-\Lambda^2)^2 + (2\sigma\Lambda)^2} \right]^{1/2} \tag{43}$$

and $$\varepsilon = \tan^{-1}\left(\frac{2\sigma\Lambda}{1-\Lambda^2}\right) + \tan^{-1}(-2\sigma\Lambda) \tag{44}$$

These functions are exactly as derived by Wheatley, ref. 4, and are shown in fig. 9.

The general feeling at BHC is that the actual craft response in sinusoidal seas would in fact lie somewhere between these two simplified solutions. This is partly due to the fact that the damping is not totally related to velocity relative to the forcing surface. Although this may be true for the cushion damping, it will not be true for external aerodynamic damping which may however be only a small part of the total.

So far, the computations and simulations performed at BHC have been restricted to consideration of the heave/pitch motion for simplicity. No cross coupling with yaw, sway or roll has been considered although in the physical model tests, the model was free to surge.

Dual Linear Spring/Damper Approach

Although the amplitudes of the motion may be appreciable in terms of craft dimensions and very large in terms of 'daylight' clearance, so that the problem is essentially non-linear, it is nevertheless informative to pursue the above approach further. A digital computer programme has been written at BHC to investigate the use of two spring/damper units, mounted in positions corresponding to the forward and aft cushion centres of a compartmented craft. No coupling between the units was assumed except via the craft hard structure and the damping was taken to be relative to the wave surface. When the results from this programme are compared with test values from a model in a towing tank with regular waves, one obtains comparisons of the type shown in fig. 10. The programme has not been fully evaluated and it would be premature to discuss the results in detail, but despite the sinusoidal input and overall linearity, the correlation with the model work is seen to be reasonably encouraging.

Digital Computer Modelling

A much more complex programme has been written for the digital computer, which takes account of the basic non linearities as far as possible.

In this programme a compartmented craft is again considered in head sinusoidal seas, although relatively minor modifications would allow any waveform to be input. The craft is assumed to have forward and aft fans supplying a bag/fingered skirt of typical BHC type. The planform is again typical of BHC craft, having a semi-circular bow and straight sides and stern. Any fan characteristic may be simulated and differential fan speeds used. Use of empirical relationships, based upon model test data, is made to account for variation of skirt geometry due to the variations in bag and cushion pressures. Iterative techniques are used to solve the equations of continuity relating the various component flows and these solutions are made simultaneously compatible with the craft attitude and accelerations at each instant. This work was done under contract for the Department of Trade and Industry and fig. 11 illustrates a typical comparison of results with SR.N4 model test values included in the final report. This slide shows that a remarkable similarity of computed and model results has been achieved, particularly with regard to bow acceleration and cushion pressure. There are however still some discrepancies in the heave and trim results. It should be noted that there are some anomalies in the model results and the waveform although regular was not sinusoidal.

When the programme was used to compute results for static forcing over a ground board, the results in table II were obtained.

Table II

| Parameter | Model | | Computer | |
|---|---|---|---|---|
| | Pitch | Heave | Pitch | Heave |
| Resonant Frequency Hz | 0.288 | 0.300 | 0.330 | 0.330 |
| Amp. magnification at resonance | 1.43 | 1.27 | 1.60 | 1.38 |
| Damping ratio | 0.375 | 0.440 | 0.330 | 0.392 |

When calculating these values, the expressions for a linear system consisting of a single spring and damper, discussed earlier, were used.

Thus, the values given are 'equivalent' values which would apply if the system were of the type postulated.

Analogue Computer Modelling

Fig. 12 shows a block diagram of the problem in a form suitable for analogue computing.

The wave is formed by a simple oscillator which has as outputs Sin wt Cos wt, w Sin wt and w Cos wt. From these components the wave height or the rate of change of wave height at any position under the craft may be determined.

The rates of change of the cushion volumes and the escape areas are sums of the appropriate wave and craft pitch and heave terms (and skirt depth).

The flow from bag to cushion is the sum of the rate of change of cushion volume and the corresponding flow out of the cushion.

The bag pressure is a function, which depends upon the fan characteristics, the flow from bag to cushion and the flow between the forward and aft bags.

The flow between the forward and aft bags depends upon the difference of the two bag pressures.

Cushion pressure is a function of bag pressure and the velocity through the feed holes.

The flows from cushion to atmosphere are functions of the appropriate cushion pressure and escape area.

Flow between the forward and aft cushion depends upon the difference between the two cushion pressures and on the escape area under the stability bag.

The craft response in heave and pitch is derived from the sum and difference of the two cushion pressures.

Many of the functions quoted above are squares or square roots. The square root terms are roots of pressure differentials which can go negative, the simulation gives a positive root if the differential is positive or a negative root if the differential is negative thus controlling the direction of flow.

The big advantage of the analogue computer is its ability to integrate in real time and to provide instantaneous solutions. Its disadvantage is that it is not good at solving complicated algebraic functions which means that many of the coefficients in the equations (inside the boxes) require detailed hand calculation.

In general the parameters which do not involve longitudinal position such as speed, wave height, weight, nominal skirt depth, feed area, etc., can be varied by simply changing one or perhaps two potentiometers, whilst the parameters involving longitudinal position such as c.g. position or stability bag position change a number of coefficients in a non-simple manner.

Simplified functions can be included in order to allow skirt depth variation with pressure ratio, but it is not practicable to try to vary skirt width (or cushion area or volume).

In order to prevent large transients the initial conditions of heave and pitch are set so that the resulting cushion pressures give zero or near zero acceleration at the start of computation. Outputs can be recorded a fixed time after the start of computation which ensures that no transients are present when recording. If necessary in order to obtain all the outputs required a case may be run many times, with the same starting conditions, and all the outputs recorded will be correctly phased. The outputs are in the form of a time history with the time base controlled by the computer.

Summarising it may be said that to carry out a parametric study by this method would involve some reasonable amount of hand calculation to obtain the coefficients, but once this was done a series of results could be obtained very rapidly indeed, provided that the loss in accuracy in the flexibility and fan flow functions compared with those used in the digital programme, is acceptable. This method could be used for a broad range of variables to highlight the areas where the more detailed digital computer solution would be of most benefit. Typical results are shown compared with model results in fig. 13.

## Hybrid Computers

The latter two computer solutions discussed above, whilst giving most encouraging results, suffer from various disadvantages, the greatest of which are:

(i) the digital programme requires lengthy computing time, but is very flexible. Most of the time is taken up during the iterative procedures, but the difficulty would be increased if the programme were extended to cope with irregular waveforms.

(ii) the analogue programme is less flexible and probably less accurate, but has the advantage that real time operation is achieved, essentially.

One can see therefore that the use of a hybrid system could combine the advantages and eliminate the disadvantages of both.

Future Use of Programmes

The above work is very recent, but its completion means that we are now in a position to investigate the effects of many geometrical and mechanical modifications to the craft currently in operation. These modifications range from investigation of the broad effects of changing 'lumped' parameters such as cushion stiffness and damping, to more detailed changes such as change of size of forward and aft compartments, and control of air flow rates between forward and aft bags and cushions. The latter is the subject of a BHC patent application which provides for complete or partial separation of the fore and aft systems, making use of non-return valves.

Finally, one might go on to consider the provision of a system, which controlled the air supply in an active manner, so that craft accelerations were minimised.

## 7. Experimental techniques

The overall range of model types and typical test programmes have been outlined in the paper on skirt design.

With regard to fan and ducting systems, model tests cover the basic impeller performance in a rig where a simple chimney intake is employed and the plenum conditions are simulated in a circular chamber of about 4 fan diameters. The pressure drops through the fan are achieved by either placing a series of gauzes in the intake, or by a variable peripheral nozzle or use of an auxiliary blower to vary pressure at plenum exit. The system performance is measured by a pitot-static grid at the intake and by pitot-static measurements near the fan exit and static pressure measurement at plenum exit. Fan r.p.m. is measured stroboscopically and power is measured by shaft torquemeter.

The overall actual craft system is measured by static hovering of a large complete model, which is fully representative of the detailed layout, including structural supports, engine intake flow and the skirt/cushion feedhole arrangement. Again, the measurements cover fan speed and power, intake pressure distribution and skirt component pressures. In some cases, fan exit pressures are measured and the plenum flow pattern is mapped out from wool tuft observations. The effects of removing plenum obstructions or of special wall shaping, or use of guide vanes, can be explored in order to investigate means for improving the overall efficiency or reducing excessive pressure gradients along the skirt duct system.

These types of measurement have also been obtained at full scale, to a limited extent. Thus, in static hover, fan speed and power, intake static wall pressures (together with 'wool tuft' observations) and skirt and cushion pressures are fairly routine measurements. More comprehensive measurements have been made by pitot/static intake rakes, together with pressure sampling at the fan exit. The salt/sand spray environment presents a problem for measurements when under way, but further instrumentation of this type is being planned for use on SR.N4. Since a full set of fan measurements is expected to require of order 100 (static and total head) pressure points at the fan eye plane (per fan), a 100 static tappings around the inlet and V-beam walls further upstream (per fan) and a further 60 static and total head pressure points around each fan exit, it will be seen that such measurement implies a sophisticated recording/sampling system. Further simultaneous measurements of this type should also be made within the rest of the plenum and skirt system, together with additional basic craft measurements. Some compromise may be necessary in practice in order to simply explore whether, for example, significant flow changes in magnitude and direction occur during motion over waves. The full measurement of basic steady system performance may then be best obtained from large scale models.

REFERENCES


| | | |
|---|---|---|
| 1 | N.E.L. Fluids Memo No. 206 June 1964 | Notes on the Design of a 24 inch Mean Impeller Diameter Mixed Flow Fan Unit for an Air Cushion Vehicle (ACV) |
| 2 | R.A.E. Technical Report 66271 August 1966 | Development of Improved Fans for the Britten-Norman CC2 001 Cushioncraft |
| 3 | N.E.L. Report No. 411 April 1969 | The Influence of Plenum Chamber Obstructions on the Performance of a Hovercraft Lift Fan. |

TABLE I

| PARAMETER | MODEL | | COMPUTER | |
|---|---|---|---|---|
| | TRIM | HEAVE | TRIM | HEAVE |
| RESONANT FREQUENCY Hz | 0·288 | 0·300 | 0·330 | 0·330 |
| AMPLITUDE MAGNIFICATION AT RESONANCE | 1·43 | 1·27 | 1·60 | 1·38 |
| DAMPING RATIO | 0·375 | 0·440 | 0·330 | 0·392 |

TABLE 1 JET THEORY COMPARISON

| THEORY | SIMPLE $k = 0.5$ | EXPONENTIAL (ELSLEY) | BARRAT | WALD – THONHOLM – HUGHES |
|---|---|---|---|---|
| $\frac{P_c}{H}$ | $\frac{2x}{1+x}$ | $(1-e^{-2x})$ | $\frac{2x}{a}$ | $1-\lambda^2$ |
| $\frac{v}{Ct}\left\{\frac{\rho}{2H}\right\}^{\frac{1}{2}}$ | $\frac{1}{(1+x)^{1/2}}$ | $\frac{(1-e^{-x})}{x}$ | $\frac{1}{a-1}\ln a$ | $-\frac{\lambda \ln \lambda}{1-\lambda}$ |
| $\frac{R}{HCt}$ | $\frac{2+x}{1+x}$ | $\frac{1}{2x}+\frac{1}{1-e^{-2x}}$ | $1+\left\{1-\frac{2x}{a}\right\}^{\frac{1}{2}}$ | $1+\lambda$ |
| $\frac{Hv}{P_c^{3/2}Ct}\left(\frac{\rho}{2}\right)^{\frac{1}{2}}$ | $\frac{(1+x)}{(2x)^{3/2}}$ | $\frac{1-e^{-x}}{(1-e^{-2x})^{3/2}}\cdot\frac{1}{x}$ | $\left(\frac{a}{2x}\right)^{3/2}\frac{1}{a-1}\ln a$ | $\frac{(1-\lambda^2)^{3/2}}{1-\lambda}\left\{-\lambda\ln\lambda\right\}$ |
| AUXILIARY EQUATIONS | $x=\frac{t}{h}(1+\cos\theta)$ | $x=\frac{t}{h}(1+\cos\theta)$ | $x=\frac{t}{h}(1+\cos\theta)$ <br> $a=\frac{1}{(x^2+1)^{\frac{1}{2}}-x}$ | $\frac{h}{t}=\left\{\frac{\lambda}{\lambda-1}\right\}\left\{\cos\theta-\frac{2\ln\lambda}{1-\lambda^2}\right\}$ |
| OPTIMUM (MIN. POWER) VALUE OF $x$ | 1.00 | $\ln 2 \doteq 0.7$ | 0.62 | 1.0 ($\theta = 45°$ only) |

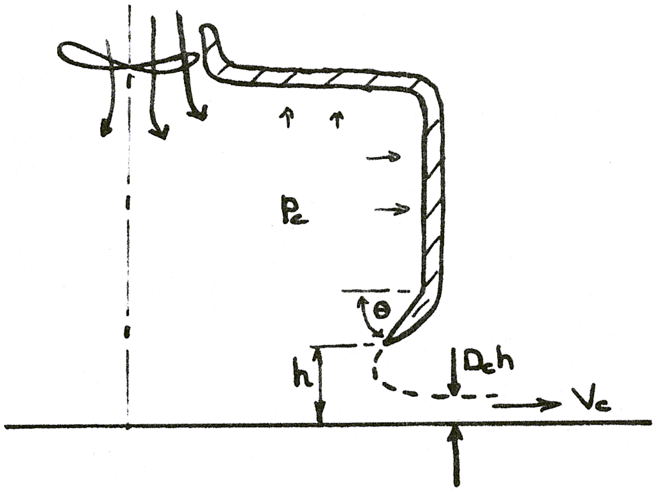



Fig.1. Simplified Section thro' Plenum Type Craft

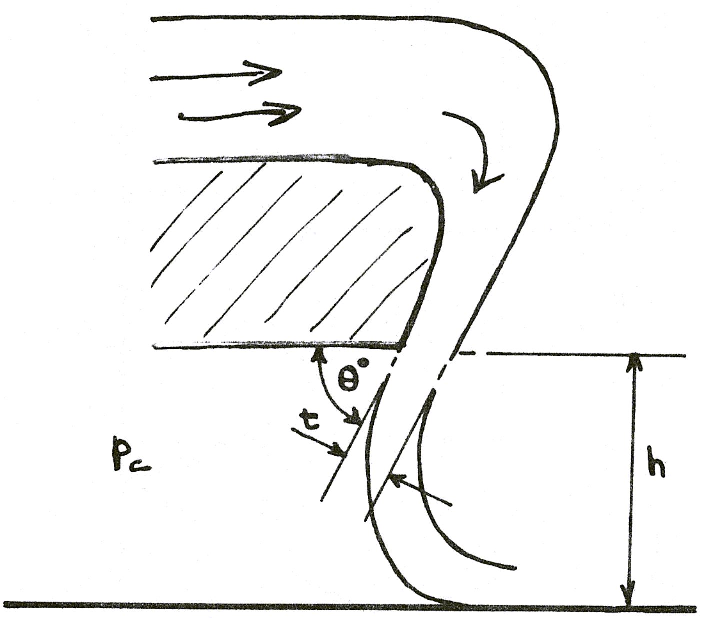



Fig.2. Simplified Section thro' Peripheral Jet Type Craft

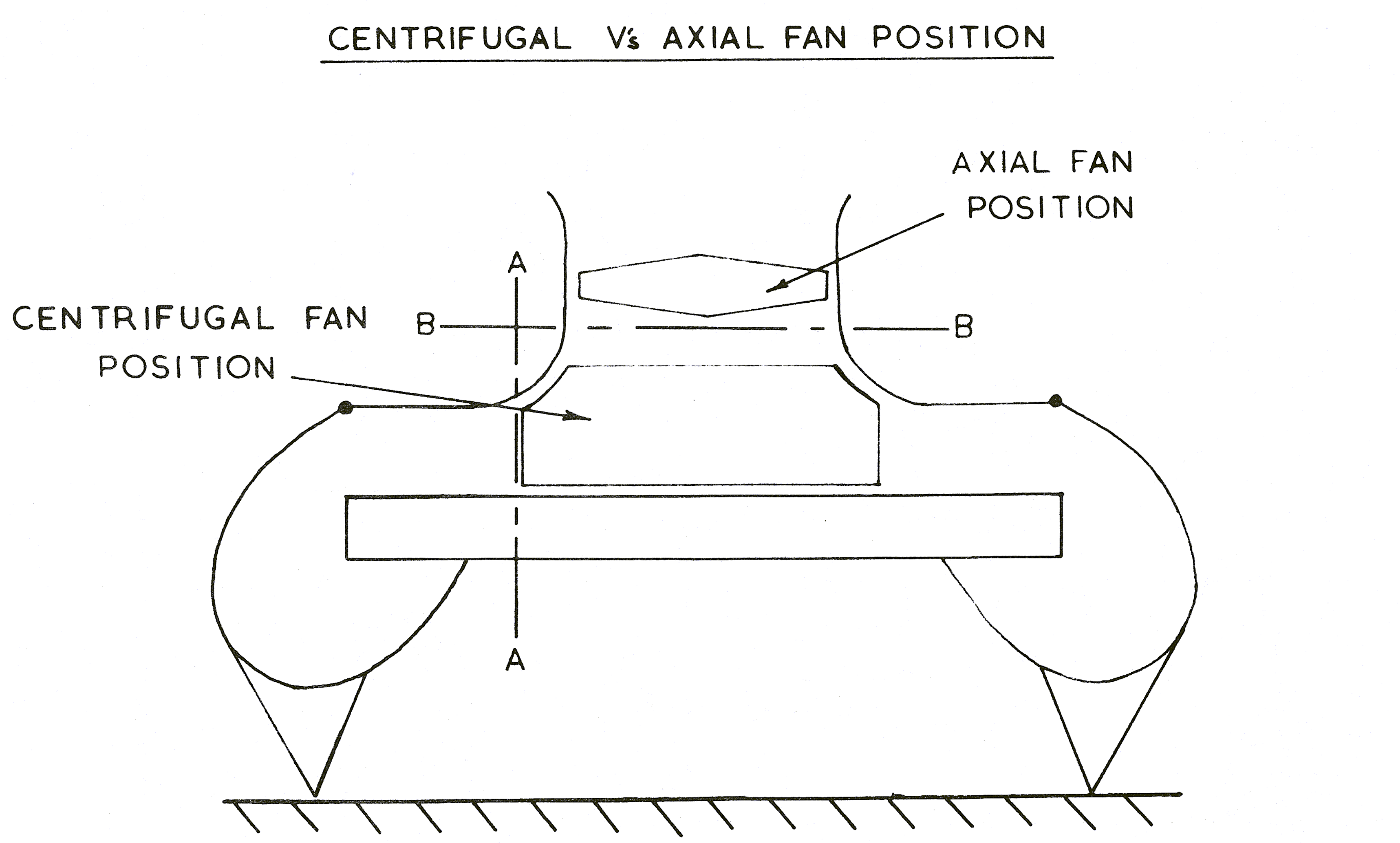

CENTRIFUGAL V's AXIAL FAN POSITION
AXIAL FAN
POSITION
CENTRIFUGAL FAN
POSITION
A
A
B
B

FIGURE.3.

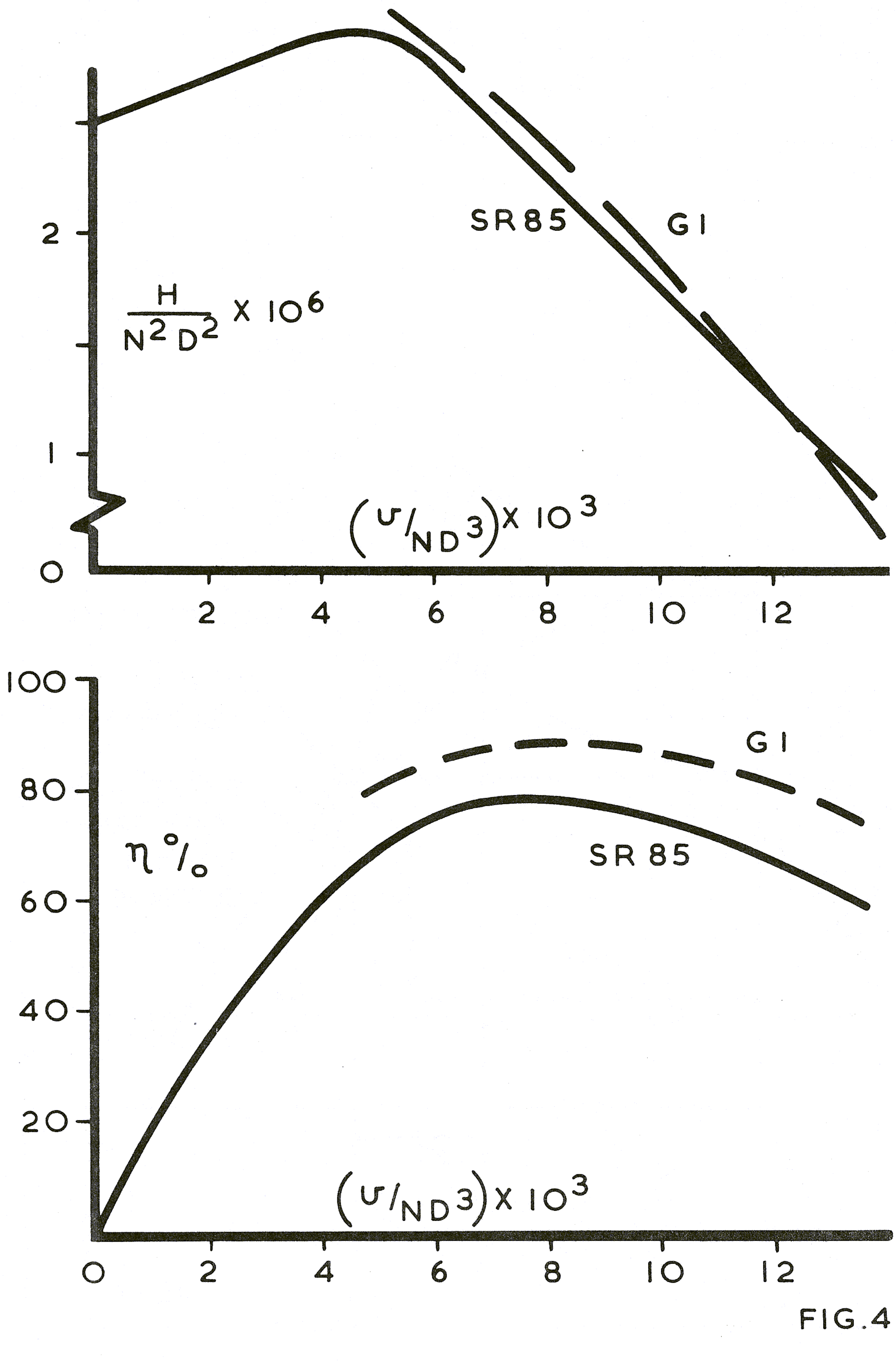

SR85
G1
2
1
0
$\frac{H}{N^2 D^2} \times 10^6$
$(v/ND^3) \times 10^3$
2
4
6
8
10
12
100
80
60
40
20
G1
SR 85
$\eta$ %
$(v/ND^3) \times 10^3$
0
2
4
6
8
10
12

FIG.4

## EFFECT OF FAN CHARACTERISTIC SLOPE ON CUSHION STIFFNESS

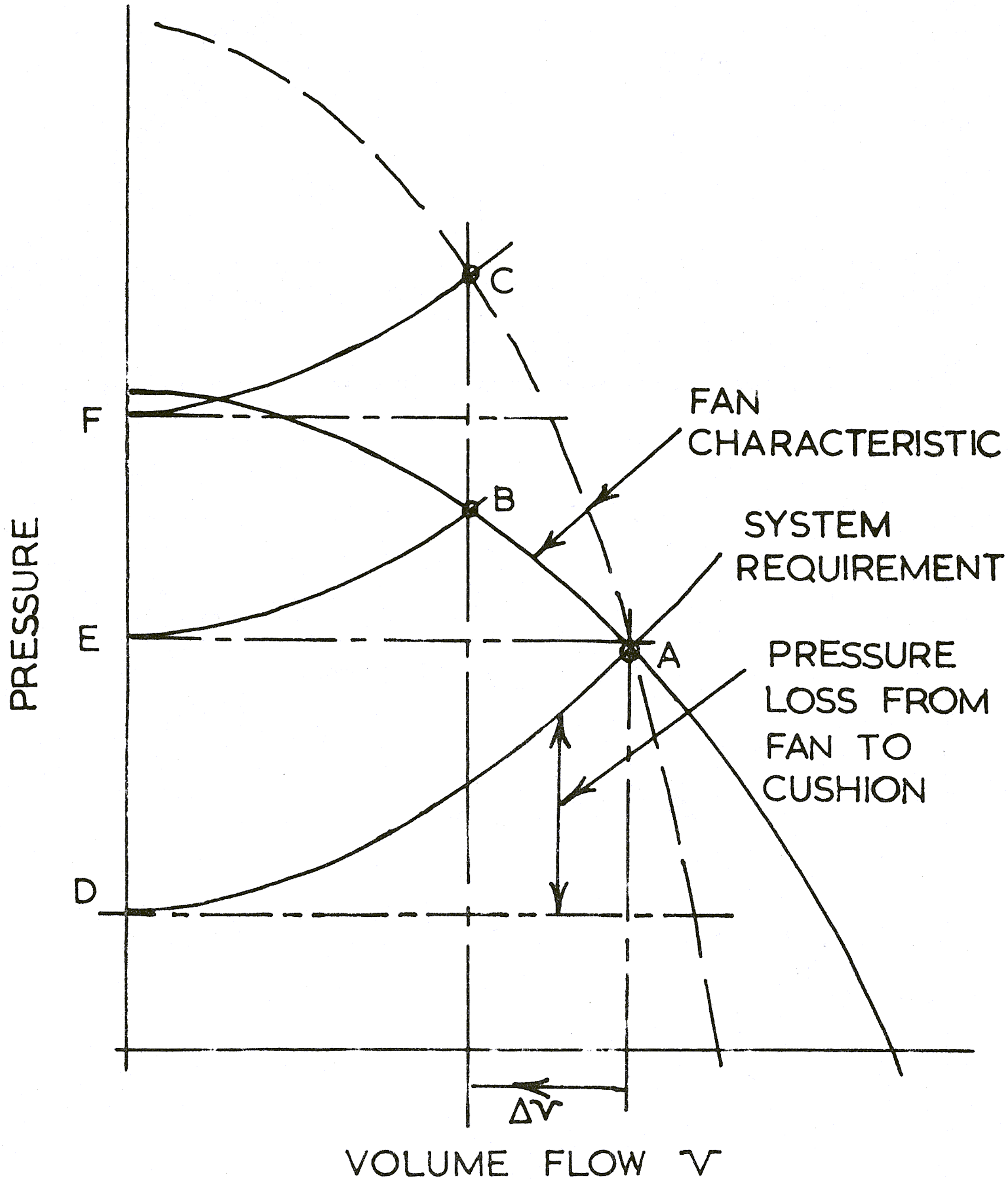



FIG. 5

# WAVEMAKING DRAG.

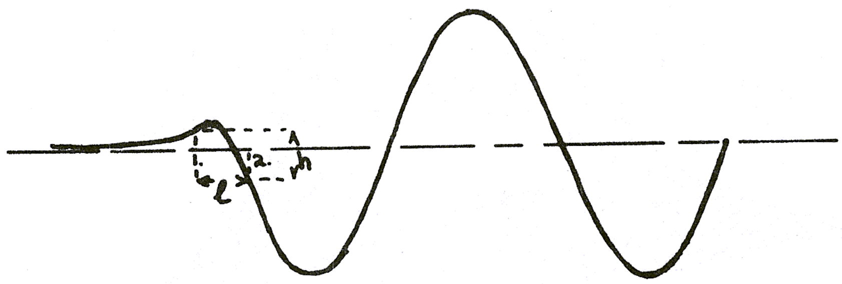



Fig. 6 Two-dimensional wave for a pressure length 1.–2. at a Froude No $\left(\frac{V}{(gl)^{1/2}}\right)$ of unity.

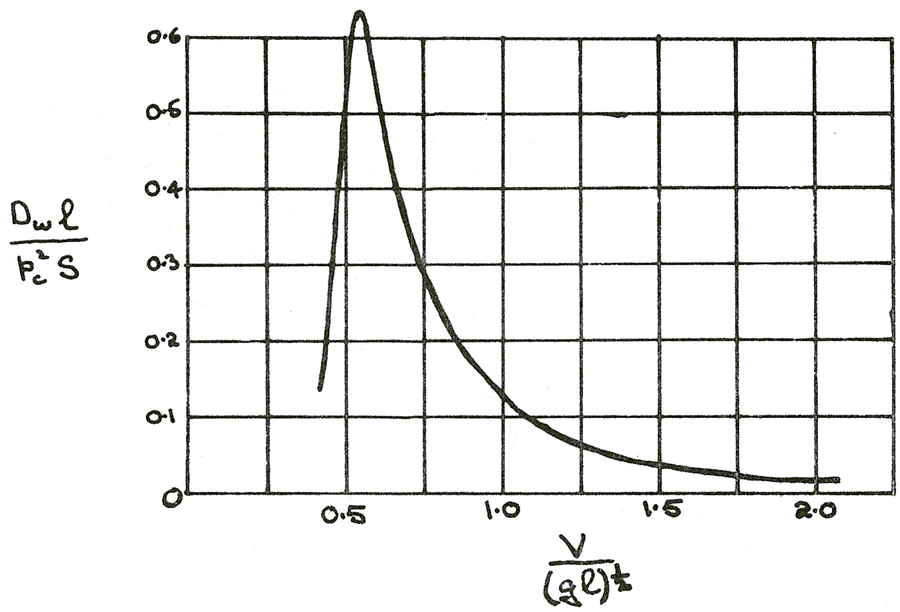



Fig. 7 Theoretical two-dimensional wavemaking drag.

based on $\frac{D_w l}{p_c^2 S} = \frac{2}{\rho_w g}\left(1 - \cos\frac{gl}{V_w^2}\right)$

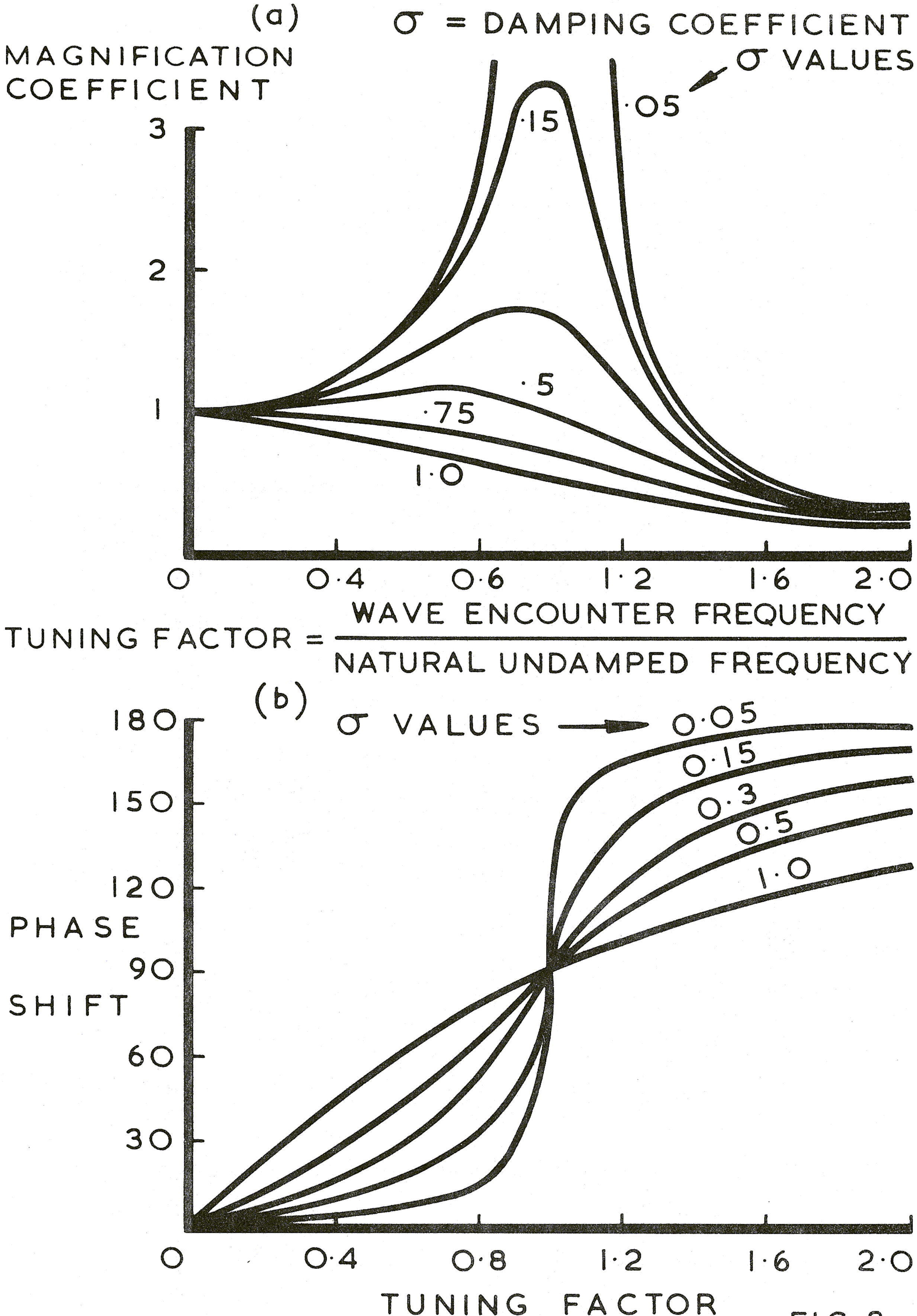

(a)
σ = DAMPING COEFFICIENT
σ VALUES
MAGNIFICATION COEFFICIENT
3
2
1
·05
·15
·5
·75
1·0
O
0·4
0·6
1·2
1·6
2·0
TUNING FACTOR = WAVE ENCOUNTER FREQUENCY / NATURAL UNDAMPED FREQUENCY
(b)
σ VALUES
0·05
0·15
0·3
0·5
1·0
180
150
120
90
60
30
PHASE SHIFT
O
0·4
0·8
1·2
1·6
2·0
TUNING FACTOR

FIG. 8

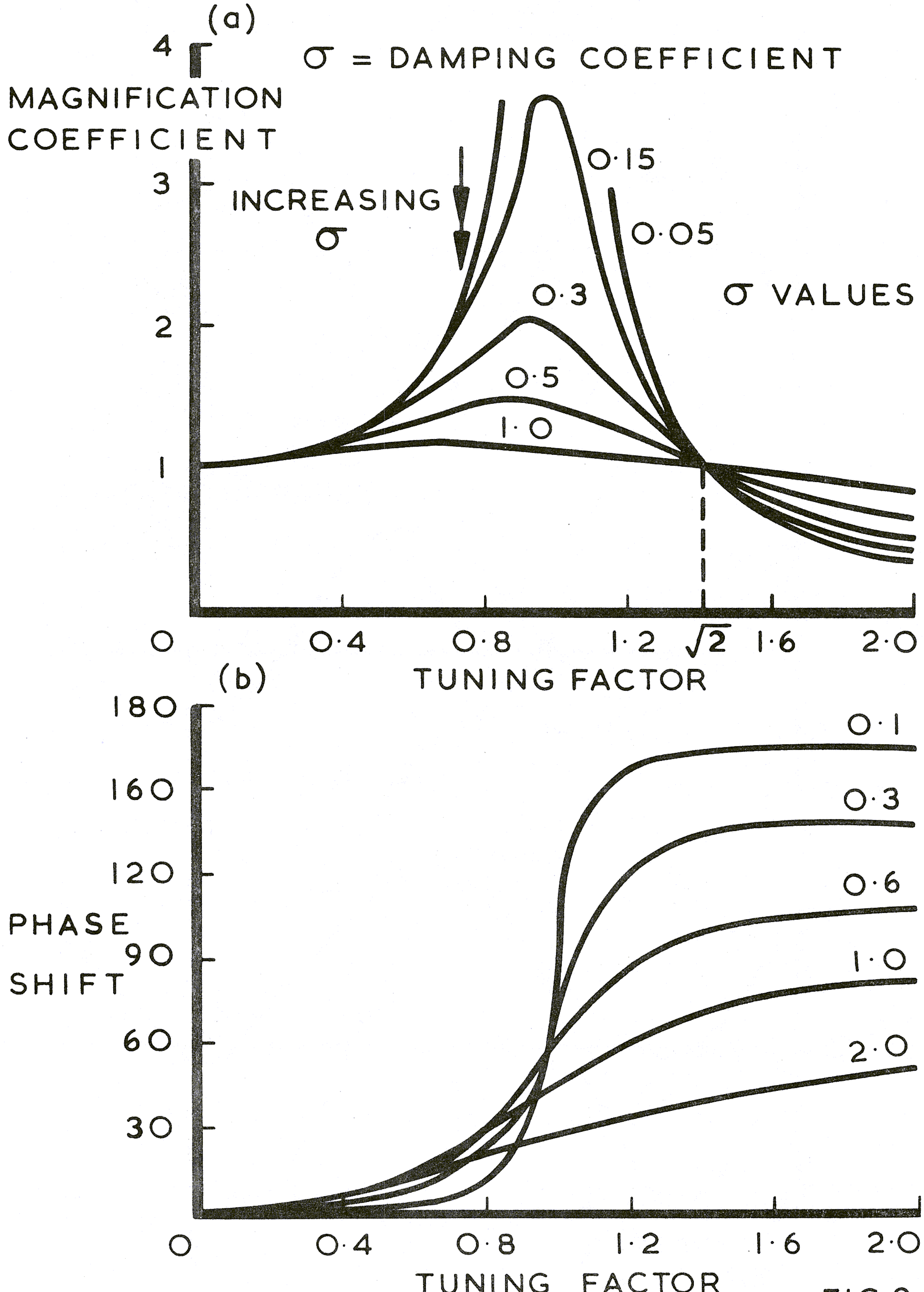

(a)
σ = DAMPING COEFFICIENT
MAGNIFICATION COEFFICIENT
INCREASING σ
0·15
0·05
0·3
σ VALUES
0·5
1·0
4
3
2
1
0
0·4
0·8
1·2
√2
1·6
2·0
TUNING FACTOR
(b)
180
160
120
90
60
30
PHASE SHIFT
0·1
0·3
0·6
1·0
2·0
0
0·4
0·8
1·2
1·6
2·0
TUNING FACTOR

FIG.9

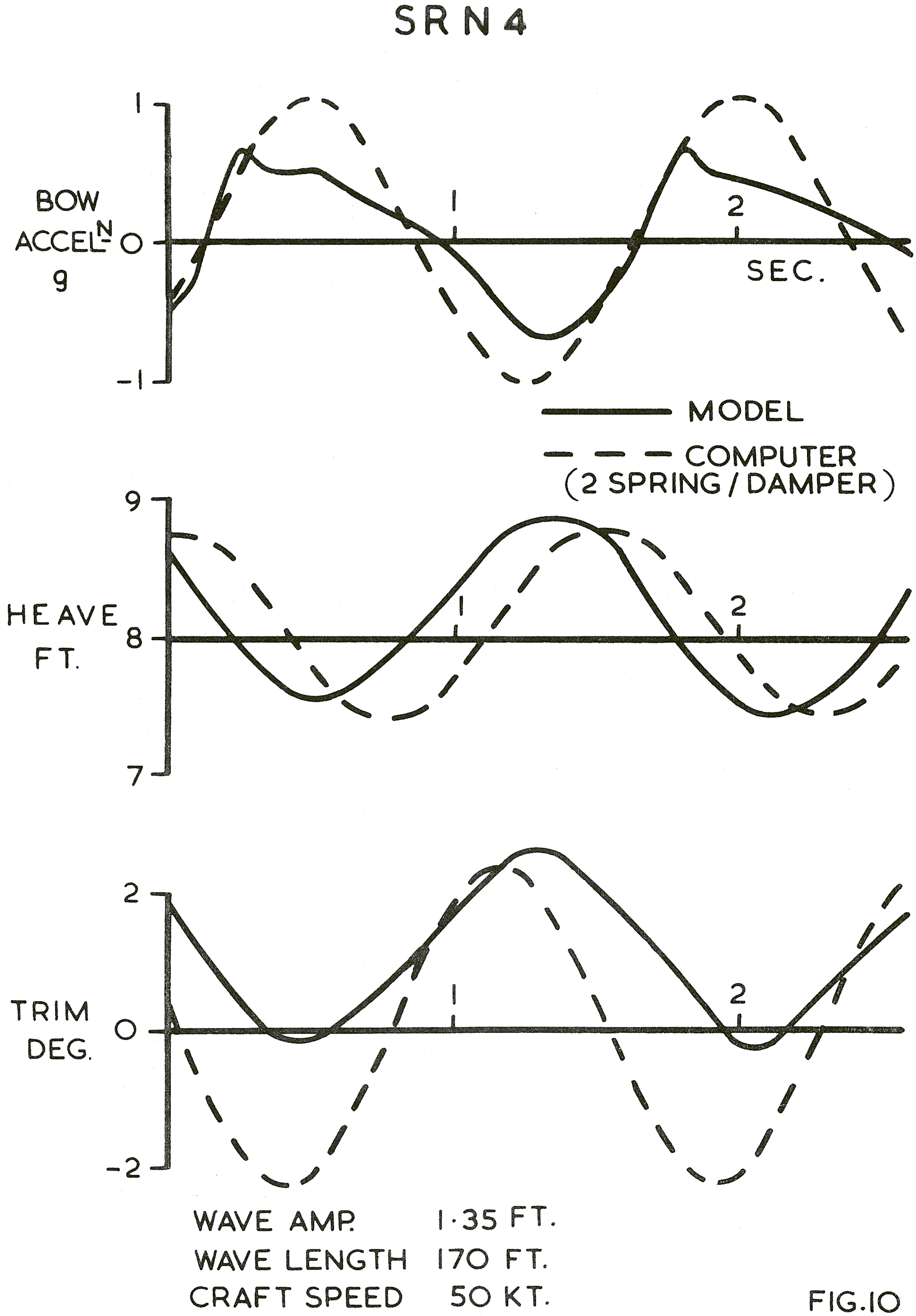

SRN4
1
BOW
ACCEL^N
g
0
-1
1
2
SEC.
MODEL
COMPUTER
(2 SPRING / DAMPER)
9
HEAVE
FT.
8
7
1
2
2
TRIM
DEG.
0
-2
1
2
WAVE AMP. 1·35 FT.
WAVE LENGTH 170 FT.
CRAFT SPEED 50 KT.

FIG.10

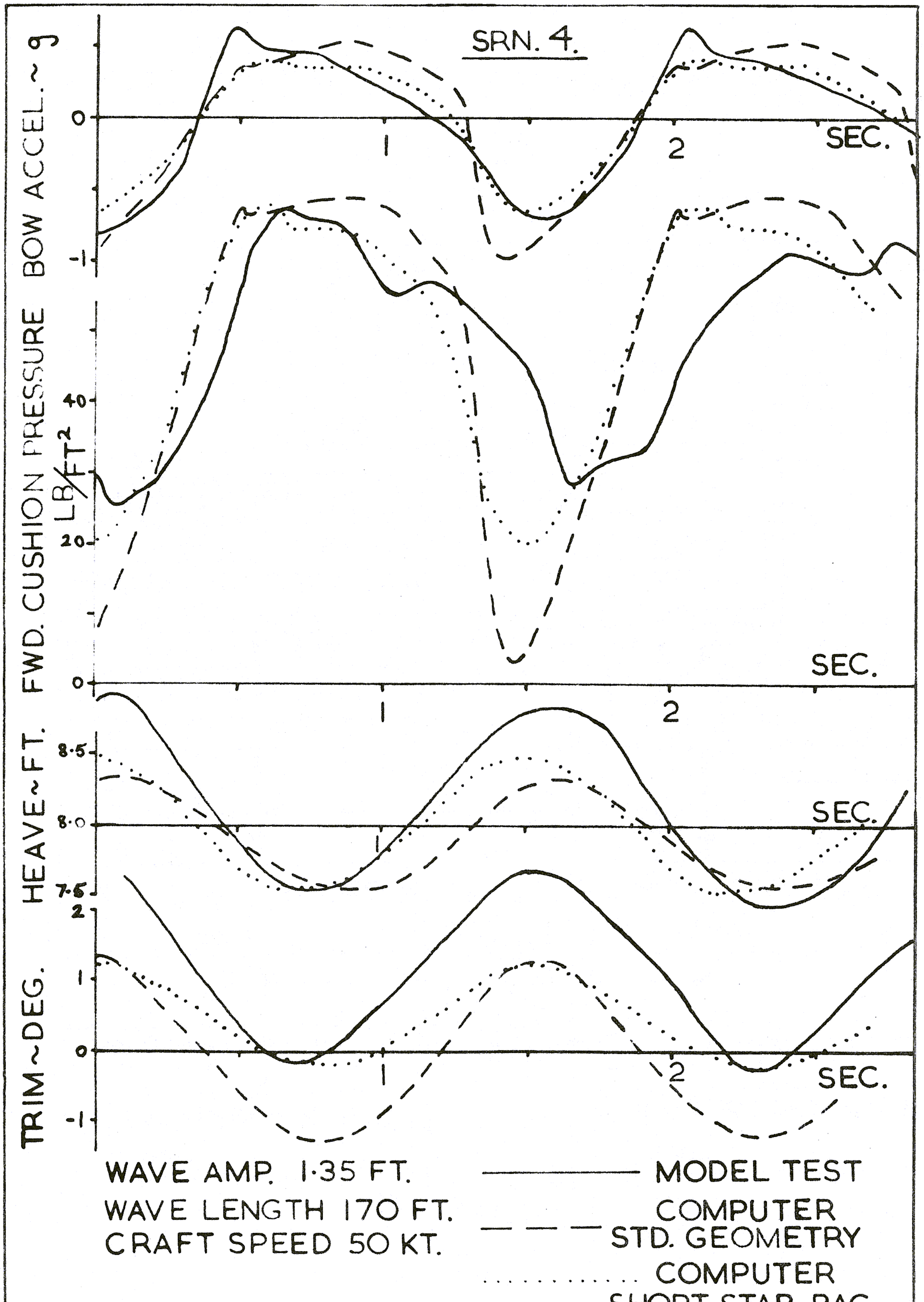

SRN. 4.
BOW ACCEL. ~ g
0
-1
FWD. CUSHION PRESSURE LB/FT$^2$
40
20
0
HEAVE ~ FT.
8.5
8.0
7.5
TRIM ~ DEG.
2
1
0
-1
1
2
SEC.
WAVE AMP. 1·35 FT.
WAVE LENGTH 170 FT.
CRAFT SPEED 50 KT.
MODEL TEST
COMPUTER STD. GEOMETRY
COMPUTER SHORT STAB. BAG

FIGURE. 11.

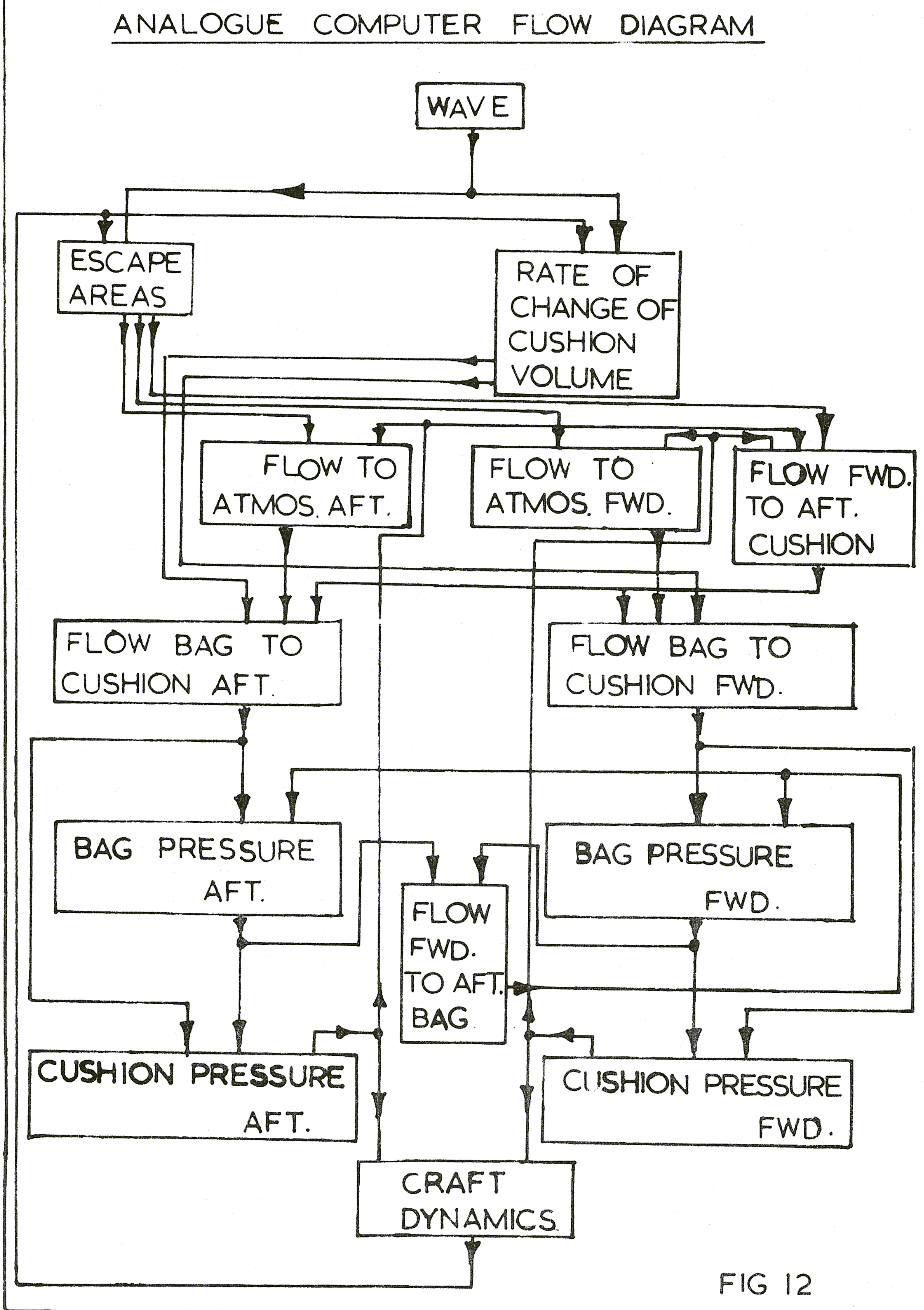
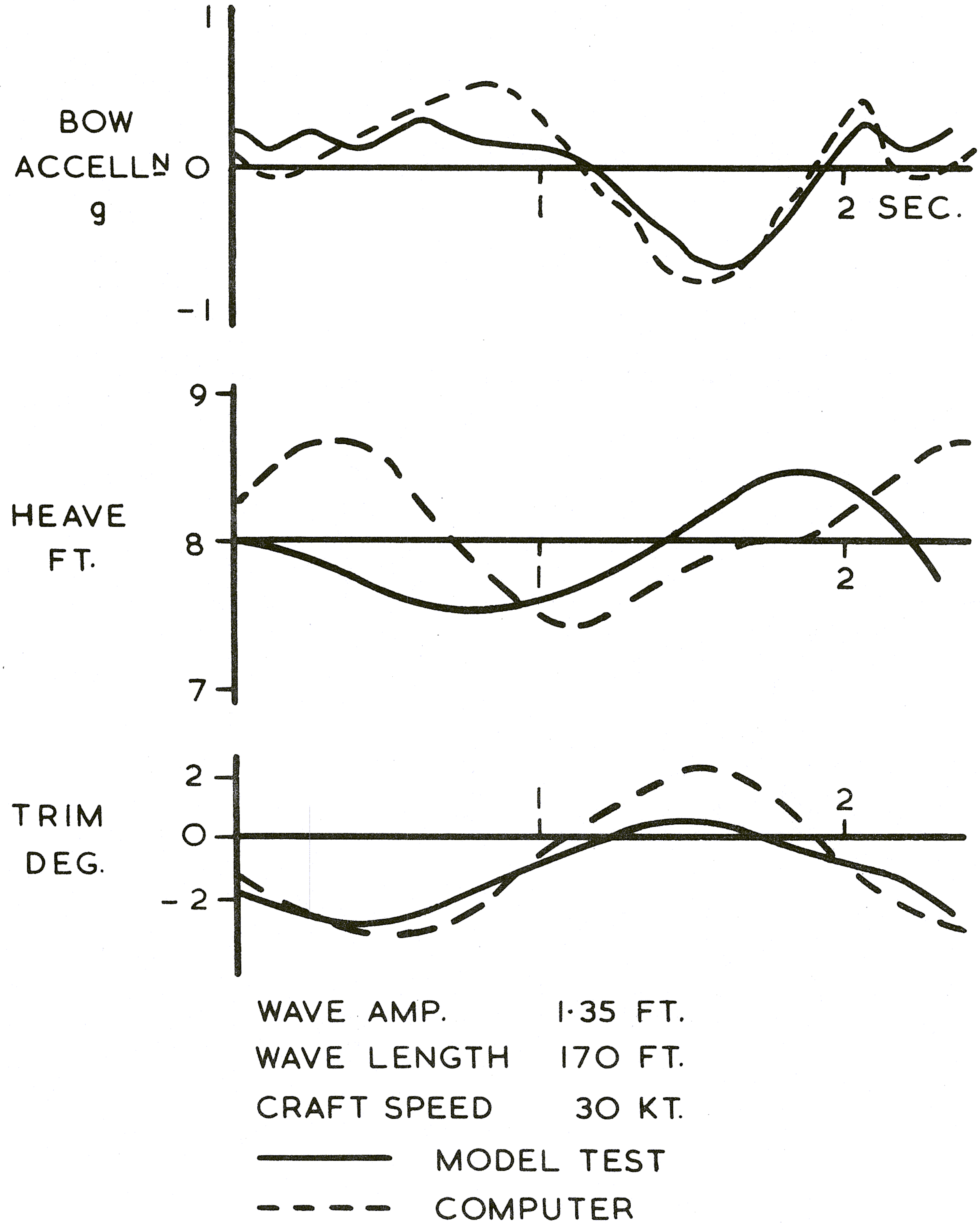

ANALOGUE COMPUTER FLOW DIAGRAM
WAVE
ESCAPE AREAS
RATE OF CHANGE OF CUSHION VOLUME
FLOW TO ATMOS. AFT.
FLOW TO ATMOS. FWD.
FLOW FWD. TO AFT. CUSHION
FLOW BAG TO CUSHION AFT.
FLOW BAG TO CUSHION FWD.
BAG PRESSURE AFT.
BAG PRESSURE FWD.
FLOW FWD. TO AFT. BAG
CUSHION PRESSURE AFT.
CUSHION PRESSURE FWD.
CRAFT DYNAMICS

FIG 12

## SRN4



1
BOW
ACCELLN O
g
1
2 SEC.
-1
9
HEAVE
FT.
8
1
2
7
2
TRIM
DEG.
O
- 2
1
2
WAVE AMP. 1·35 FT.
WAVE LENGTH 170 FT.
CRAFT SPEED 30 KT.
MODEL TEST
COMPUTER


FIG.13